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THE INFLUENCE OF COMPUTERS ON COMPRESSOR TECHNOLOGY

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ABSTRACT

Computer contributions to the advancement of compressor technology are deemed to have commenced with the presentation of 4 papers on simulation models (by Wambsganss & Cohen, Touber, Najork, MacLaren and Kerr) to the IIR Congress at Madrid in 1967. The paper outlines the history of developments since then of computer simulation models of positive displacement compressors with which the author has been concerned. These related to improving the mathematical description of the valves and of the pipework system in which the compressor operates and evolving faster and better numerical methods to solve the equations. It is suggested that the combination of such models with optimisation techniques is at a very early stage of development but will eventually become a routine procedure, with the digital computer being used as part of the only practical way to arrive at a "best" design. It is forecast also that, as computer capabilities continue to increase, more detailed studies of valve durability under bending and impact stresses will become economically feasible, the addition of more science to the art of valve design being considered to be necessary.

The separate role of the computer in controlling high speed data acquisition systems is discussed; such systems can acquire, process and output experimental data taken during a compressor cycle with great ease and accuracy. Such data are necessary to assess the validity of the predictions by simulation models and optimisation techniques. Advances in such data acquisition systems have been so rapid in the last decade that the best available at the time of the first Purdue Compressor Technology Conference (1972) is now obsolete.

The paper concludes by referring to the problem of the assimilation by the compressor designer of the rapidly growing volume of new information being generated by these computer applications. Acknowledgement is made of the significant contribution made by Purdue University in providing an appropriate forum for the dissemination of this information.

INTRODUCTION

If the time span of known usage of compressors could be compressed into one calendar year then the air bellows used for metallurgical processes in China and Egypt would be dated as 1 January and today would be 31 December. On this time scale the application of mechanical power to drive compressors by steam engines did not commence till mid December. From that development, less than two centuries ago, there has been an ever increasing addition of science to the ancient art and craft of compressor design and manufacture. The growing understanding of thermodynamic principles in the mid-nineteenth century led to the introduction of multi-stage compression with intercooling. By the end of the century the economies to be achieved by increased speed and smaller machines led to the almost universal adoption of the self-acting compressor valve. That component, simple in concept but with an infinite variety of geometry and complexity of behaviour in operation, is often considered to be the Achilles heel of the reciprocating compressor. Leonardo da Vinci, that genius of so many arts and sciences, appreciated some of the desirable features of automatic valves exactly 400 years before the founding of the Hoerbiger Valve Company in Vienna in 1896. The developments by Ing Hanns Hoerbiger date from the beginning of the last week of our hypothetical year of compressor history.

The trends in design apparent at the end of last century continued through the first half of this century. About the only analytical procedure to predict compressor power and capacity was that based on the pressure-volume diagram shown in Figure 1. It is assumed that the valves have infinite lift and flow area so that there is no pressure difference required across them during the flow processes in the suction and discharge phases of the cycle.

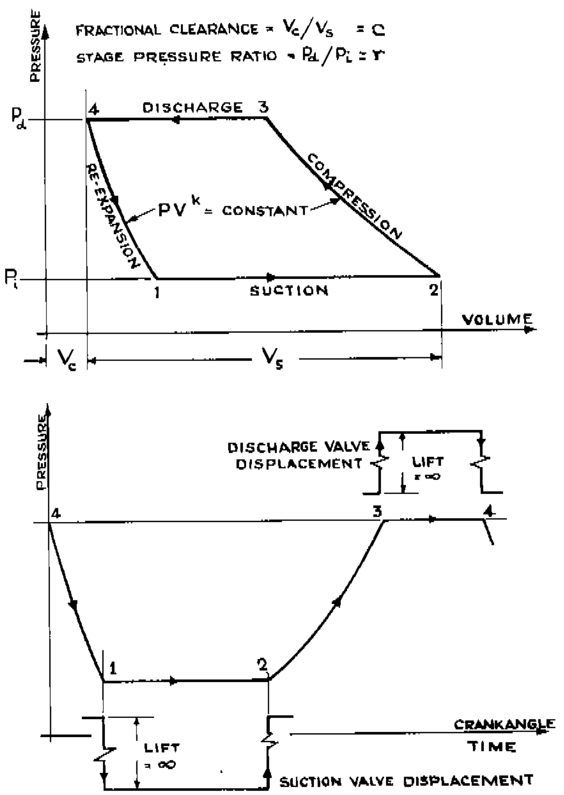


Fig. 1 Pressure-volume, pressure-crankangle and valve displacement diagrams for a compressor with hypothetical valves.

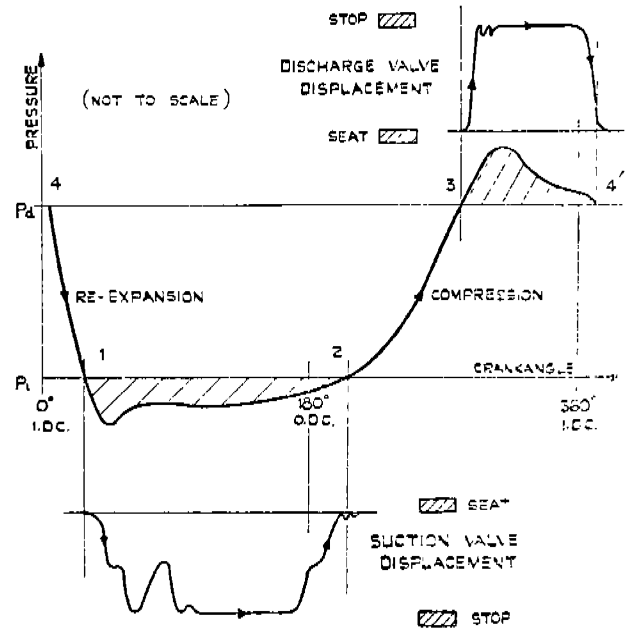


Fig. 2 Pressure-crankangle and displacement of automatic valves, assuming that plenum chamber pressures (P_i and P_d) remain constant

Generations of engineering students have deduced for a compressor stage that:-

$$\text{Indicated Power} = \frac{k}{k-1}\, mRT_2\, (r^{(k-1)/k} - 1) \text{ and}$$

$$\text{Indicated Volumetric Efficiency} = 1 - c(r^{1/k} - 1)$$

This simple analysis accounted for about ten of the many variables to be selected by the designer: it provided no information whatsoever about the compressor valves.

Around 1940-1950 an advance in analysis permitted a computation of the pressure/time (or crankangle) history in a compressor cylinder together with an account of the behaviour of the self-acting valves. Hence it became possible to estimate the loss in power and capacity due to the valves (indicated by the shaded areas in Fig.2).

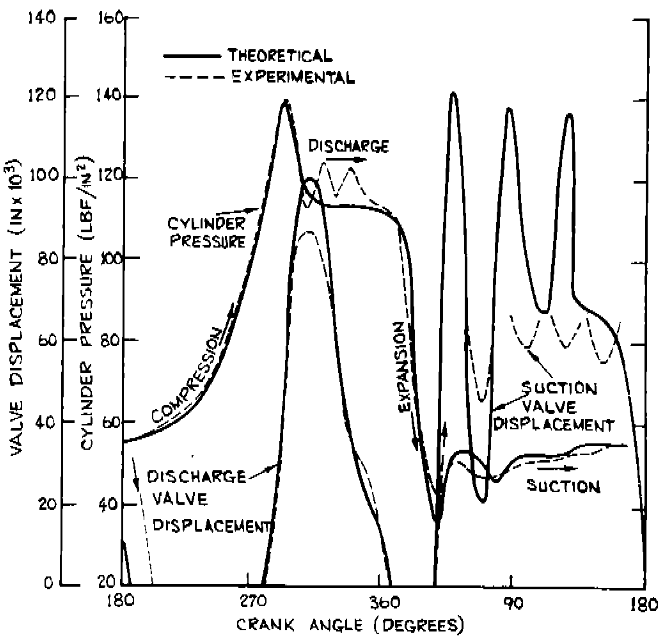
This advance was claimed by Russia on account of the publications by Dollezal (1) around 1940: similar independent work by Costagliola (2) was reported in the USA in the late 1940's. In the DSc thesis submitted by Costagliola to M.I.T. there was no reference to the earlier work of Dollezal, probably because of difficulties in international communication during the second world war. My intention is to outline the contributions made during the recent relatively short period since the publications of Dollezal and Costagliola. Only during the last 20 years, following the general availability of computers, were these contributions of practical use to the designer. Hence this review is concerned mainly with events in the last day or two of our hypothetical year of compressor history, ie the starting date is about 27 December. The advances in these last few days have been so rapid that one has difficulty in prophesying what developments the near future holds in store.

MATHEMATICAL MODELS OF A RECIPROCATING COMPRESSOR WITH VALVES

Although the automatic valve is simple in principle, the mathematical description of its behaviour in its environment is not. The valve lift is an unknown function of the unknown pressure difference across the valve and hence dependent on compressor speed, valve mass, spring stiffness, etc. A mathematical description of the cycle depicted in Figure 2 can be considered to consist essentially of two simultaneous differential equations, a "flow" equation which expresses the flow through the valve as a function of the pressure difference across the valve during its operation and a "dynamic" equation which expresses the displacement of the moving element in the valve in terms of the forces acting on the element due to this pressure difference. Both equations are expressed as functions of time and hence bear a simple relationship to the crankangle of reciprocating compressors, particularly if uniform angular velocity can be assumed: incorporated within the mathematical description is a relationship between crankangle and piston displacement. This relationship between angle of rotation and volume displacement can be determined for all types of positive displacement compressors, sliding vane, single or twin screw etc: the geometric relationship may be more complicated but the volume/angle of rotation relationship is determinable. When the stroke is not fixed, eg in the free piston compressor, the mathematical model becomes more complex.

By present day standards the models proposed by Dollezal or Costagliola were crude, containing many simplifying assumptions. Nevertheless they raised the number of variables which could be accounted for from approximately 10 (Figure 1) to about 50 (Figure 2). However the procedures available, by graphical methods or mechanical calculators, to solve the non-linear simultaneous differential equations proposed by Dollezal or Costagliola were so time consuming and the results so inaccurate that their mathematical models were of little practical interest to the designers of that time. This situation was changed dramatically by the advent of the computer which permitted the rapid and cheap solution of the equations which were the basis of such mathematical models. The year 1967 marked this breakthrough when four papers (3,4,5,6) were presented to the XII International Congress of Refrigeration held in Madrid. The authors were Wambsganss and Cohen, Purdue University, USA; Touber, Delft University of Technology, Holland; Najork, Institute Luft-und Kaltetechnik, Dresden, E Germany; and MacLaren and Kerr, University of Strathclyde, Scotland. All had utilised a computer to solve equations similar to those developed by Costagliola. It is notable that all four papers came from academic institutions, reflecting the fact that such centres had access to computers for the development of the necessary programs earlier than did most industrial research establishments.

Each of these investigators claimed that the theoretical predictions of compressor and valve behaviour obtained from his model showed sufficient agreement with the corresponding experimental results over a range of operating conditions that the model could be used with some confidence to study a design and so indicate the likely consequences of any proposed modification to it. The model could make such predictions quickly and cheaply whereas the study of the many compressor and valve variables experimentally is slow, expensive and frequently impractical. For example, it is difficult to instrument and to change variables independently on, say, a 3000 horse-power oxygen compressor and difficult, for other reasons, to accommodate even miniature sensors in a small domestic refrigerant compressor. Results from an early attempt at the latter exercise were described by Wambsganss and Cohen to the Congress at Madrid, a sample result being shown in the now familiar Figure 3.



HERMETIC COMPRESSOR: 1/4 hp ; 3600 REV/MIN ; R12 ; SUCTION PRESSURE 55·7LBF/IN²; COMPRESSION RATIO 2; INDEX n = 1·12

Fig 3 Wambsganss and Cohen (Ref 3). Comparison of prediction by simulation model with experimental records.

As a consequence of the difficulties associated with and the expense of conducting experimental investigations, computer programs for mathematical models are now widely used to simulate compressors of various types and sizes, pumping air, industrial gases and refrigerants. While increasingly sophisticated models have been and continue to be developed, many industrial organisations have found that relatively simple models often suffice. The simpler the model the more economical it usually is in terms of computer time. Hence a criterion of usefulness is that the model should be as simple as is consistent with providing an adequate answer to the particular questions being asked of it by the designer. The questions may be such that the complexities of valve behaviour can be neglected (as for example in the models developed by Kruse (7) for calculating piston and bearing loads), that the working fluid be can be treated as a perfect gas, that cyclical heat transfer effects are small, that pulsations in the flow into and out of the compressor can be neglected, etc.

MODEL DEVELOPMENT SINCE 1967

Many simplifying assumptions have to be made when constructing any mathematical model of such a complex physical situation. A continuing aim is to reduce the number and restrictive nature of these assumptions in order to create still more accurate models.

It was assumed in early models that flow through the valves could be described by conventional one dimensional nozzle theory with inclusion of appropriate empirical flow and force coefficients. These coefficients were assumed to be constants or to vary with valve lift and empirical values were obtained from steady flow tests. In operation a compressor valve has opened and closed again before steady flow can be established hence these steady flow test results, obtained from rigs in many laboratories round the world, are suspect. However to establish "dynamic" coefficients of flow and force is difficult: Brown and Fleming will be discussing this topic at the present Conference. In future it is likely that their experimental study of dynamic flow in valves will be undertaken, not by miniature hot wire probes as at present, but by computer controlled laser anemometry which does not interfere with the flow. Others have ambitions to estimate such coefficients by calculations using appropriate sub-routines in the programs for the overall simulation model so that a design could be studied prior to any manufacture. An attempt to apply that philosophy was made by Hamilton (8) in a paper to the 1972 Conference.

UNSTEADY GAS FLOW

One of the simplifying assumptions made in the model relating to Figure 2 is that pressures in the plenum chambers, P_i and P_d, remain constant during the suction and discharge phases of the compressor cycle. The gas flow through a positive displacement compressor must be intermittent and, since a valve plenum chamber and its associated pipework have finite volumes, the pressures therein must vary. The magnitude of the variations will also be influenced by the changing piston velocity during the suction and discharge process, and the effective flow area through the valve (which varies with valve lift which in turn depends on both cylinder and plenum chamber pressures). Hence the pressure-time history in the plenum chambers, P_i and P_d is complex, as illustrated in Figure 4, and affects the pressure in the cylinder and the motion of the valve. A much more complex model is required to account for these gas pulsation affects.

Figure 5 illustrates that such a model can predict the marked difference in compressor capacity (about 15%) which occurred by merely changing the length of the inlet pipe (expressed as a "delay angle") to an air compressor. Pulsations can be sufficiently severe in practice to cause valve flutter, valve slamming, valve failure, or vibrations which may lead to excessive noise and even failure of pipe-work.

Probably the first to extend a model to account for pulsation effects was Brablik (9) at the CKD Compressor Factory in Prague, Czeckoslavakia. He described some of his work to the 1974 Conference. Soedel at Purdue University also contributed much to this subject.

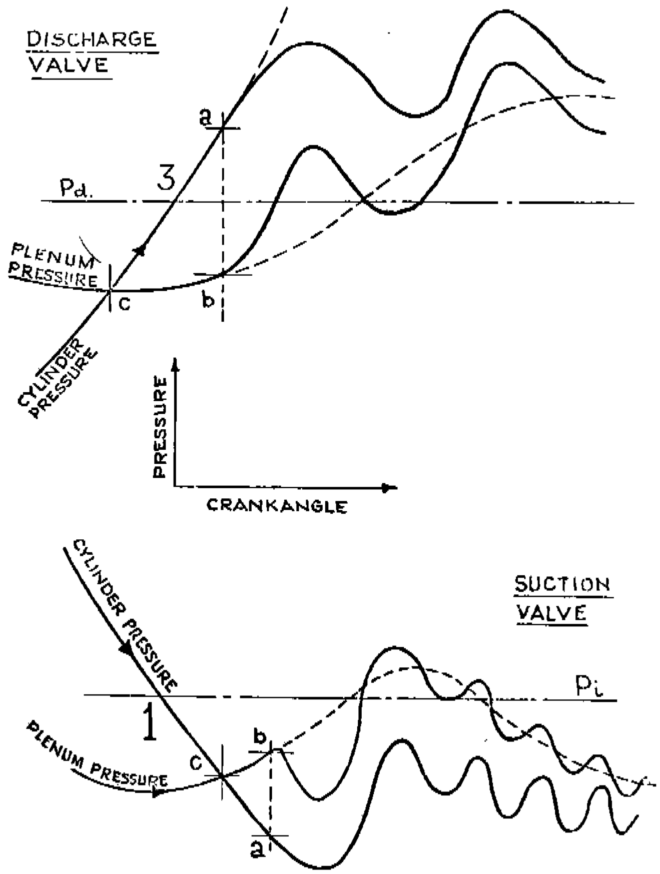


Fig 4 Effect of piston velocity and valve opening on cylinder and plenum chamber pressures.

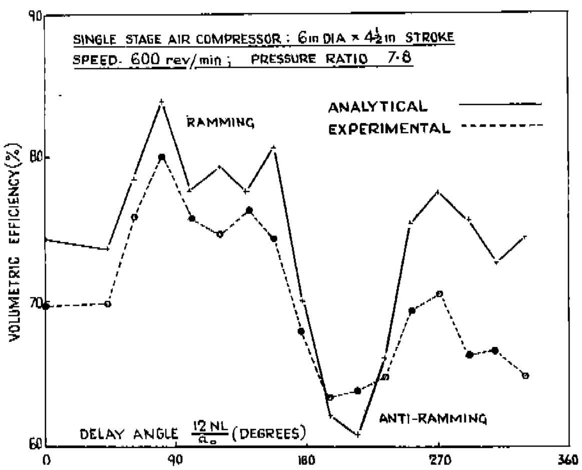


Fig 5 Comparison of predictions by a simulation model, which accounts for flow pulsations, with experimental records of volumetric efficiency.

Both Brablik and Soedel assumed that pulsations were of sufficiently small amplitude that the "acoustic" equation could be applied without incurring significant error. The first model to simulate a compressor system accounting for finite amplitude pulsations was that by the late Professor R S Benson and Ucer (10) at the University of Manchester Institute of Science and Technology, England. They assumed the flow to be homentropic (although the effect of friction in the pipes was allowed for) and solved the hyperbolic partial differential equation which describes one dimensional unsteady flow in a pipe by the method of characteristics. Tramschek, (11) at the University of Strathclyde, Scotland, removed the assumption that the flow was homentropic. In addition to considering the flow to be non-homentropic, which allowed heat exchange in an interstage cooler to be accounted for, he applied improved numerical methods to solve the flow equations. These methods were not only more economical of computer time but provided more accurate solutions by allowing the amount of damping in the equations to be specified. A description of cavities, dampers, changes in pipe cross-section, pipe junctions, etc have all to be available in the model in order to determine, for example, whether a pressure or a rarefaction pulse is in the plenum chamber during the short time when a valve is opening or closing. Procedures to handle these boundary conditions were developed by Tramschek (12) and reported in two papers to the 1976 Conference. An extensive experimental program was undertaken to assess the validity of the simulation of the many boundaries which may be encountered. Figure 6(A) illustrates a few of these discontinuities but under the heading of dampers alone there are many configurations and types (resistive, reactive and non-reactive) to be considered. Figure 6(B) illustrates predictions by his model of the cylinder and plenum chamber pressure variation and valve displacements for a single stage air-compressor accounting for the effect of the pipework system in which the compressor was operating.

Figure 7 allows an assessment to be made regarding the extent of the agreement obtained between such predictions and experimental records obtained from a two-stage intercooled air-compressor. These tests showed that the gas column in the simple inter-stage cooler employed was subjected to two forcing pulsations: a compression pulse generated during the discharge of the L.P. stage and a rarefaction pulse generated during the suction process in the H.P. stage. Hence the amplitude and phasing of the pulsating flow in the intercooler and the phasing of the crankangle between the two stages could have a significant effect on the suction process in the H.P. stage, particularly on the displacement of the H.P. suction valve. At present Tramschek is investigating the interaction between the three cylinders in parallel of a single stage semi-hermetic compressor pumping R12, similar to earlier work by Benson and Ucer, and by Soedel.

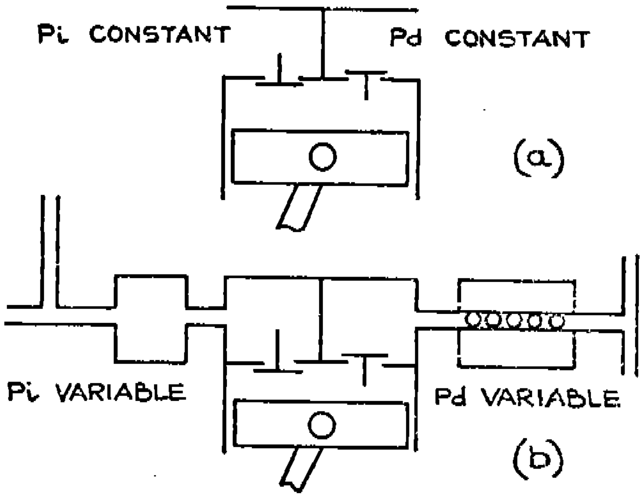


Fig 6(A) (a) Compressor in isolation
(b) Compressor in pipework system.

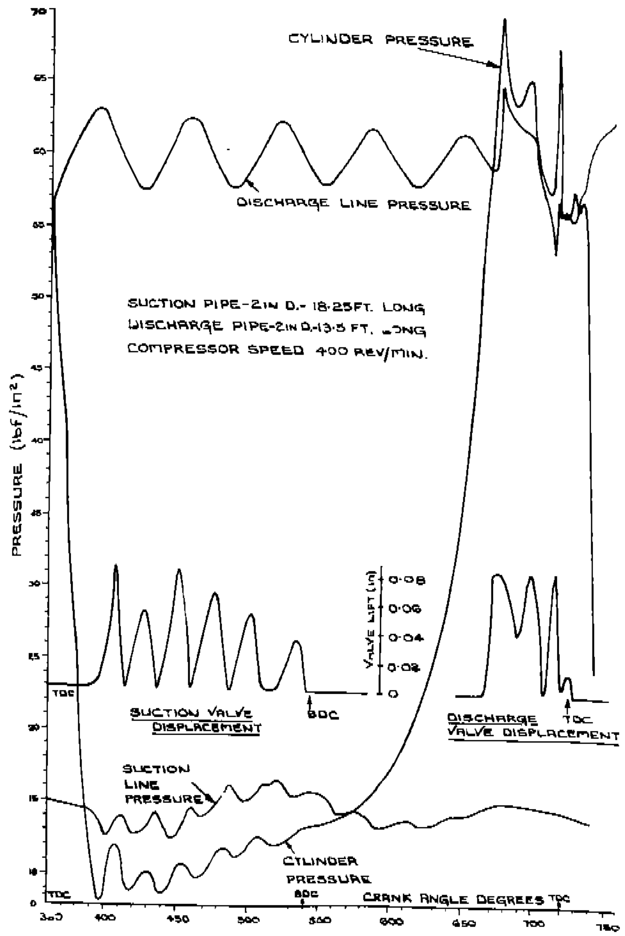


Fig 6(B) Computed pulsations in cylinder and plenum chambers, and valve displacements for compressor in a pipework system.

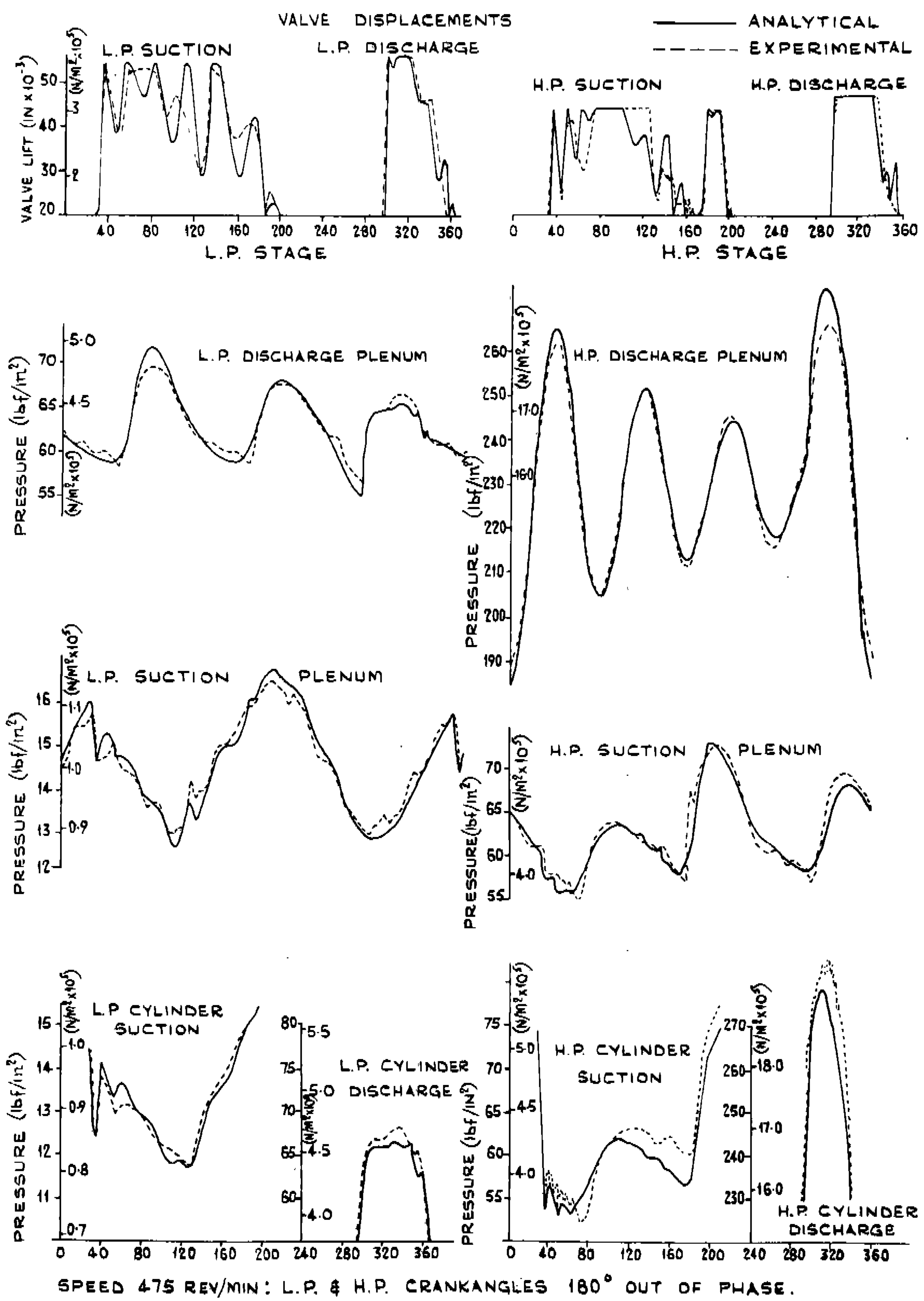


Fig 7 Comparison of predictions by simulation model, accounting for flow pulsations, with experimental records for a two-stage intercooled air compressor.

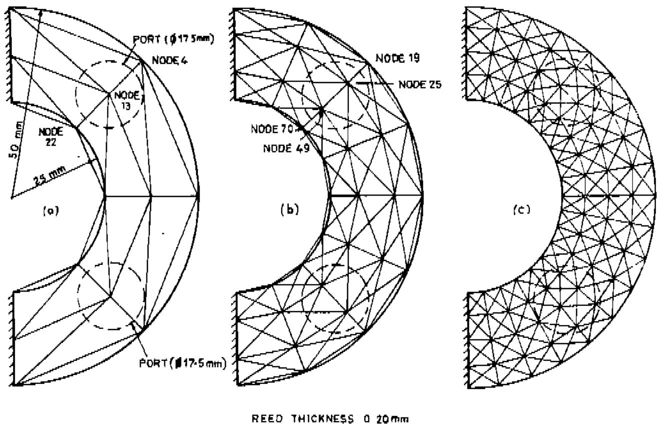
In most models to date it has been assumed that the pressure distribution within the cylinder is always uniform: as the speed and capacity of computers increase this simplifying assumption will be less necessary: the computation of velocity and pressure distribution in engine cylinders is an active field of research at present. It will also be necessary to predict more accurately the pressure-time histories across valves. These histories must be known precisely in order to arrive at good estimates of the velocity of the moving element of a valve at impact on stop and seat. Only then can reliable predictions be made of values of dynamic stresses in valves due to bending and impact.

The time and cost of developing such models to describe both the compressor and the pipework systems within which the compressor operates and with which it interacts prompted a number of users and manufacturers to co-operate and to form, some 25 years ago, a "Pipeline and Compressor Research Council" under the auspices of the Southern Gas Association. The Southwest Research Institute at San Antonio, Texas, was contracted to develop methods for pulsation prediction, suppression and control. Large analog computers were dedicated to this purpose. This facility was discussed by the director of S.W.R.I.,Von Nimitz (13) at the 1974 Conference.

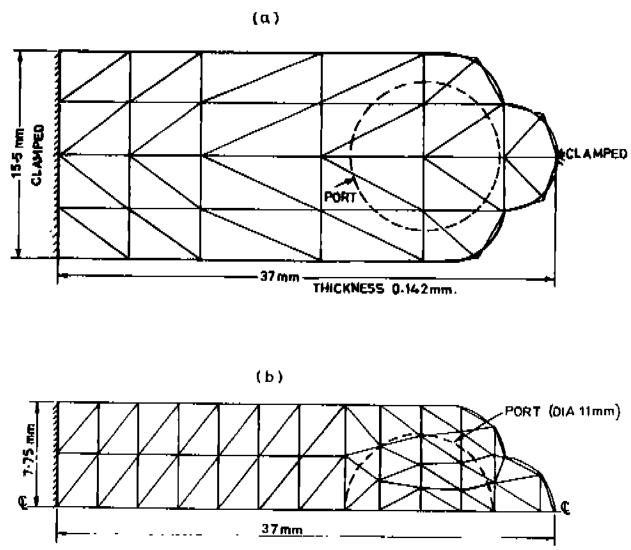
VALVE DURABILITY

There have been several incentives to develop mathematical models of reciprocating compressors. Initially an improvement in compressor performance was sought by using such analytical studies to reduce losses in compressor power and capacity due to the valves. This first objective has been met to a great extent as a consequence of the development of simulation models since 1967. Another objective has been to develop analyses which would determine the dynamic stresses in the valves due to bending and impact, that is, to add some science to the rather black art of valve design: a century of practical experience of valve failures has not provided an empirical panacea to the problem of valve durability. Cohen (14) inaugurated discussion of valve stress analysis at this Conference in 1972, and Soedel (15) contributed an early paper in 1974. At each Compressor Conference held at Purdue University since then there has been an increasing number of papers relating to the topic.

A powerful technique, the finite element method, has been increasingly applied during this period. The method, providing sufficient computer facilities are available, permits calculation of dynamic bending and impact stresses in valve reeds and plates. Again Purdue University made an early contribution with a paper by Hamilton (16) in 1976. While most reports on the subject have come from academic sources, it is refreshing to observe that at the Conference in 1980 a major contribution on this important topic came from manufacturers of steel in Sweden. Advances continue and several papers on the subject and the technique will be read before the present Conference. The validity of the technique to evaluate dynamic bending and impact stresses, and thereby permit an estimate of valve durability to be made, has been established. The finite element grids used, such as in Figure 8, (A) and (B) are generated by the computer at the outset. However the grids shown are relatively coarse because the procedure is very demanding of computer time and storage when the dynamic (as distinct from the static) situation is being examined.



Number of Elements: (a):16 (b):96 (c):288
Number of Nodes : (a):15 (b):36 (c): 95
Number of Degrees of Freedom : (a):27 (b):84 (c):255
Fig 8(A) Finite element models of a half-annular reed valve.



(a) 85 Degrees of Freedom (Whole Reed)
(b) 114 Degrees of Freedom (Half Reed)
Fig 8(B) Finite element models of a cantilever reed valve.

Figure 9 illustrates the maximum dynamic stress along the centre line of a cantilever reed valve using two grid sizes. In this particular case the maximum dynamic displacement of the reed at the centre of the port was about 50% greater than the permitted lift of the reed at the tip. No doubt the ever increasing capacity of digital computers will permit more refined solutions to be obtained in future at reasonable cost.

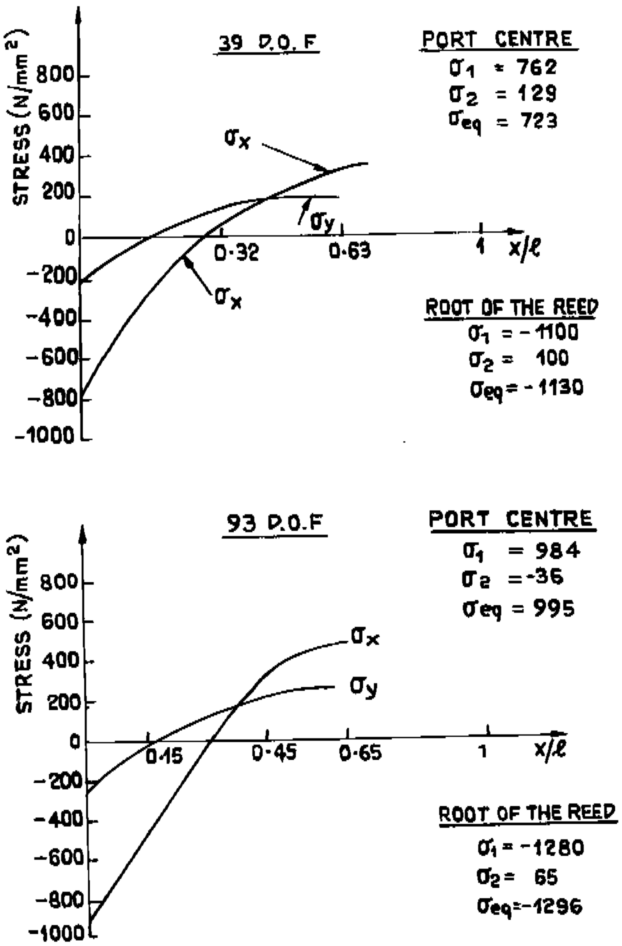


Fig 9 Maximum dynamic stress distribution along centre line of a cantilever reed valve, using two sizes of finite element grids.

OPTIMISATION OF A DESIGN

If a mathematical model adequately simulates a compressor then, as mentioned earlier, an aid is available to the designer to predict cheaply and quickly the likely consequence of changes in the many parameters at his disposal. His objective is to arrive at the "best" design for the specified duty (which may include a range of steady state operating conditions, transient and part load conditions and various refrigerants). To do so experimentally is quite impractical. It is not possible to study the variables independently and it is too difficult and expensive to measure the small latent improvements perhaps possible by a change of each variable individually or to arrive at the optimum combination of them. It is possible to use the mathematical model alone to study the multitude of combinations of various values of the relevant parameters and then select the "best" combination, but this somewhat random approach is prohibitively expensive in computer time and cost. However, an appropriate optimisation technique can handle the variables involved in an orderly and economical manner and arrive at values which in combination will give the "best" design. In delegating this exercise to a computer responsibilities for making engineering decisions are not abrogated - in fact these responsibilities become more onerous. The "objective function" which is to be optimised has to be specified and this is not as easy as it might appear. In addition any practical design is subject to many "constraints" and these have to be identified and quantified at the outset. While some constraints can be readily specified (eg mains frequency and motor speed) values for others are not readily available (eg tolerable valve impact velocity), so demands on the skill and experience of the designer are increased.

An early application of this philosophy was made at Purdue University when, in the last session of the Conference in 1974, 5 papers were presented, all reporting work at Purdue University. An initial impetus had been imparted by Ragsdell during his short stay at Purdue University. At the next Conference, in 1976, Hamilton (18) presented a paper which reflected the interest at that time in the design of sliding vane compressors, stemming from earlier work by Coates (19). However this early impetus was not maintained, the number of papers appearing at the Purdue University Conferences under this subject heading being 5 (1974), 2 (1976), 1 (1978), 2 (1980). This lack of growth is surprising since the coupling of optimisation procedures to mathematical simulation models appears to offer a powerful aid to the designer, particularly since there is no practical alternative way to achieve a "best" design.

Plastinin, from the Bauman Higher Technical University in Moscow, in the later part of his keynote address to the Compressor Conference at Purdue University in 1978 indicated that the philosophy was being applied in Russia. He reported a study which led to the determination of the optimum bore to stroke ratios for the stages in a reciprocating compressor.

In reports by Kerr (20) to recent Conferences the objective functions were defined as forms of thermodynamic efficiency for the valves (in 1976 and 1978) and for the compressor (in 1980). A relatively simple simulation model of a reciprocating compressor conforming to Figure 2 was used with a modified form of the "Complex" optimisation procedure. In the first study the number of variables allowed to float was 12 (6 dimensions for each valve), under one specified operating condition of an air compressor fitted with single ring-plate valves. Figure 10 shows that this optimisation procedure interacting with the simulation model converged to optimum values of these 12 parameters, and hence to the maximum objective function, after only approximately 210 iterations. By so modifying the 12 valve and spring dimensions a compressor efficiency of about 10% above that of the design in production was predicted. The variation in the three cases A,B,C, in Figure 10 resulted from changing the constraint of the upper limit of what was considered to be a tolerable value for valve impact velocity on the valve stop, emphasising the comment made earlier that the designer must not only specify the practical constraints but also quantify them.

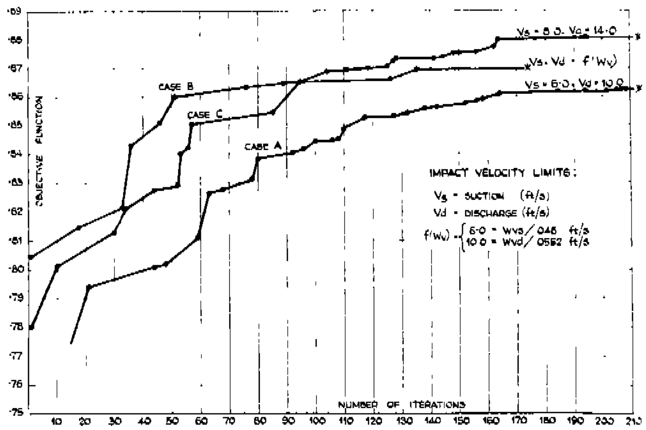


Fig 10 Optimisation of objective function (thermodynamic efficiency) of air compressor with 12 valve dimensions as variables.

In most studies to date a "best" compressor has had the connotation of best thermodynamic performance: these studies have been concerned with parameters such as power and capacity. In a PhD thesis by Hoare (21), supervised by Kerr, and recently accepted at the University of Strathclyde, an attempt has been made to include some economic considerations together with thermodynamic factors in arriving at a "best" design. Even so, this potentially powerful design procedure is still in its infancy.

VALIDATION OF MODELS

When assessing the validity of the predictions from a simulation model the prime concern is interpretation of the inevitable difference (hopefully small) between the predicted values and experimental records. As stated earlier, the simulation model is an approximate description of the complex physical situation: simplifications must be made if the mathematical model is to be capable of solution. The experimental records contain experimental errors: it is necessary to expend resources of money and time to minimise these errors. Preferably all the experimental measurements should be made during one compressor cycle and the digital computer has permitted this to be accomplished. The computer serves as a controller, data processor and memory bank and it has peripherals to serve as graphical or digital output devices.

An early computer controlled high speed data acquisition system, developed by Kerr (17), was based on a Hewlett Packard 2100 mini-computer which had 16K memory. A schematic diagram is shown in Figure 11(A).

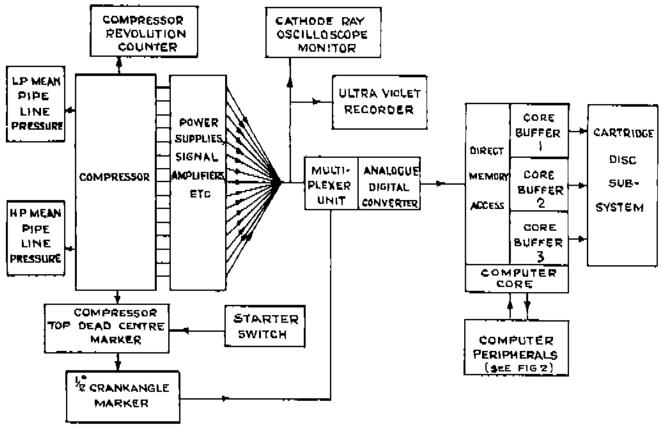


Fig 11(A) Instrumentation and control configuration of high-speed data acquisition system (1972).

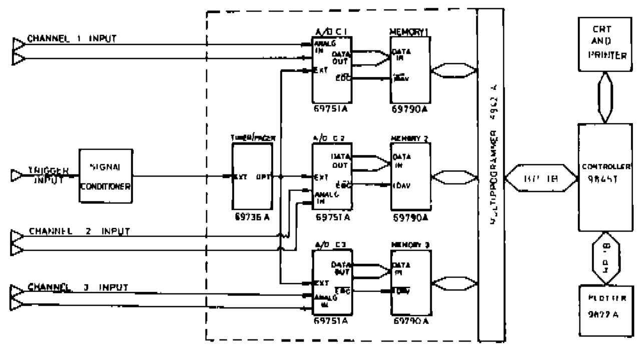


Fig 11(B) Three channel high-speed data acquisition system (1982).

The analog signals from the transducer in each of up to 16 channels were conditioned and converted to digital form in the analog-digital converter (ADC) which had a maximum processing rate of 100,000 signals per second. The system was used to acquire the experimental results shown in Figure 7 taken from a two-stage intercooled compressor. The pressures at the various locations were sensed by piezo-electric transducers and valve displacements were monitored by inductive transducers. In addition a phase-marker was necessary and a trigger pulse was used to instruct the system to acquire data sequentially from the channels at pre-selected crankangle intervals (say ½°). The control programs, written in Assembler, made provision for two methods of operation. In the first method, buffer stores were provided in the computer core. One buffer stored incoming data from the ADC while, simultaneously, data previously acquired into another buffer was transferred to store on a cartridge disc sub-system. This procedure allowed data greatly in excess of the computer core capacity to be acquired during a test by re-using core which had been allocated as a buffer store following the dumping of its previous contents onto the disc. In the second method all the data was stored in core. This permitted a somewhat higher rate of acquisition by eliminating the time to execute the instructions in Assembler for the transfer of data into the disc store but restricted the storage capacity to about 14K of the computer memory. When processing the data the computer, in which the calibration of the transducers had been stored, provided a linear output, and graphed results to pre-specified scales for ready comparison with predicted results from a simulation model. Figure 7 provides an example of the superimposed results.

In addition to acquiring, processing and outputing experimental results for the compressor during a cycle, as in Figure 7, part of a compressor cycle could be examined in detail. Figure 12 illustrates the use of this facility to examine the displacement of a single ring plate suction valve: the time between readings was 10 μs but only each second reading (at 20 μs intervals) is plotted on Figure 12.

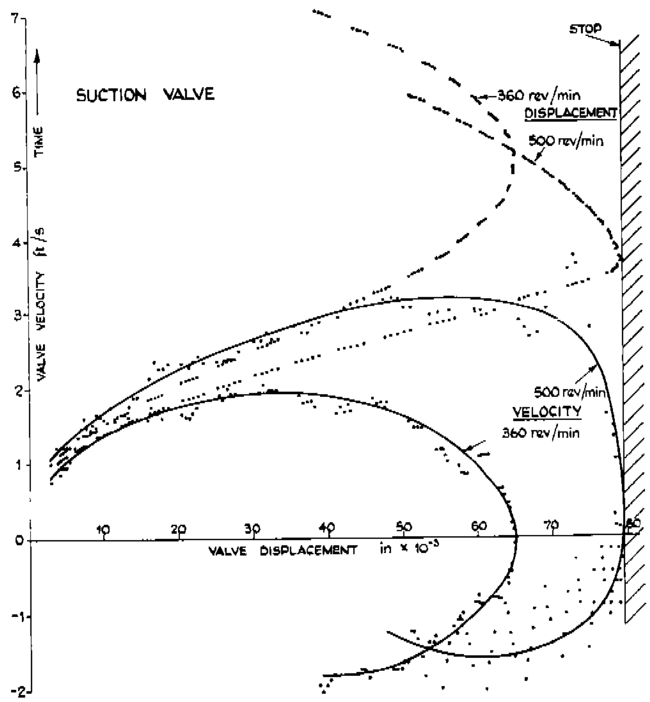


Figure 12 Measured displacement and velocity of ring plate suction valve during opening.

Such detailed experimental valve displacement and velocity diagrams are important in studies relating to valve behaviour and durability. For example, the experimental evidence in Figure 12 confirmed that the simulation model could predict well the velocity of the valve, during most of its lift. However, the model did not provide information relating to the "squish" damping on the moving element of the valve as it approached the valve stop. The experimental evidence would permit an appropriate deceleration due to the "squish" effect to be included in the model as an empirical correction. As a consequence of this "squish" effect the effective impact velocity at the valve stop is less than the impact velocity predicted by a model in which the damping effect is not accounted for.

The computer system described above must operate in real time and consequently requires a fast computer to control the acquisition of data during a cycle in a high speed compressor. The installation, with its peripheral devices, was purchased in 1973 and was expensive and bulky. Its operational system was sophisticated: this coupled to a lack of mobility tended to result in it becoming dedicated to a particular application.

The more recent advent of microprocessors and ADC's with inbuilt digital stores allows the use of small desk top computers which are mobile, easy to program and operate, and are cheaper. Figure 11(B) illustrates such a system. Each signal channel is allocated an individual ADC which has a capability of 33000 readings per channel per second backed up by a digital store of nominal 4K capacity. Thus a total acquisition rate of 33000 N readings per second is available, where N is the number of channels. In the earlier system the maximum total acquisition rate of 100 000 readings per second permitted 100 000/N readings per channel per second.

The newer system, commissioned in 1982, is initiated as before by an appropriate trigger input signal and thereafter the ADC's are triggered simultaneously by the timer/pacer unit (at time intervals equivalent to ½ degree of rotation or crankangle interval, say). The analog readings are converted to digital signals and stored in the memory card. The stored data can then be presented to the computer sequentially, at a relatively slow time rate, for conditioning, processing and output in digital or graphical form. Thus real time computer operation is avoided and higher acquisition rates achieved at low cost. This system will be described in more detail in a paper to the present Conference: it is being employed at present at the University of Strathclyde to study experimentally the compression process in a cell of a sliding vane oil-flooded air compressor during one revolution of the rotor.

THE PLACE OF THE COMPUTER

The advent of the computer has sparked off an explosion of activity in the field of compressor technology. The solution of mathematical models of compressors (including the systems in which they operate) and the assessment of the validity of the predictions from the models by comparison with accurate experimental records, has only become a practical procedure because of the availability of computers. The computer also facilitates the retrieval from libraries of the new information which is being amassed.

The duration of this explosion, a mere 16 years since the IIR Congress in Madrid in 1967, ie in the last two days, 30 and 31 December of the hypothetical calendar year, is obviously short in relation to the time during which man has made and used compressors. Computer capacities will increase so we are only at the beginning of a long and interesting road to be trodden by both academic and industrial researchers. It has given satisfaction to the academics to observe the rapidly increasing adoption of this form of computer aided design by their colleagues. Advances will further assist the designer who will be required to produce ever more efficient, reliable and cheap compressors, spurred on by market forces, the high cost of energy, and government legislation demanding higher efficiencies. Increasing computer resources will help to offset the disproportionate effort which will be required to win smaller and smaller design improvements in absolute terms. For example, in a new range of hermetic compressors for air conditioning applications an increase of performance is claimed from 9.5 Btu/W to 10 Btu/W, over that of the previous range: the development was aided by "extensive computer modelling of design options". Obviously, it will be more difficult to achieve the next 0.5 Btu/W improvement. To this end, systematic optimisation into a "best"design of the many variables involved should become a standard procedure. Such a procedure is only possible with the computer but much remains to be done to effectively implement this philosophy.

COMMUNICATION

There is too much new information being generated for it to be condensed into a text-book. Perhaps this explains the shortage of modern texts on compressors. The excellent text of earlier years by Frenkel (22) published in Leningrad in 1960 was never translated into English. The text by Chlumsky (23) published in Prague was translated into English in 1965 but that also predates these developments in computer aided design. The books by Frenkel and by Chlumsky are both out of print. Plastinin (24), who prepared the keynote address to this Conference in 1978, has recently published a small book (178 pages) which gives a fuller review of the topics discussed in this paper. The book by Plastinin contains a valuable list of 273 references - valuable because the author, having been the Scientific Secretary to the several large Compressor Technology Conferences held in Russia since 1967, is in a unique position to assess the papers published in both the East and West. It remains for someone to translate Plastinin's book into English.

While such books provide an overview of the subject, we must turn to published papers for detailed information. It is in this connection that our host, Purdue University, has given a great boost to the advancement of Compressor Technology. By providing an international forum each second year since 1972, the University has assisted the promulgation of recent developments by pre-Conference short courses and by publishing more than 350 papers in the Conference Proceedings. Purdue University is personified in this context by Professors Ray Cohen, Jim Hamilton and Werner Soedel who have provided much of the new information by their own researches, and have arranged this forum yet again for us in 1982. I am sure that we will make good use of this opportunity provided by them during the three days of the present Conference.

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